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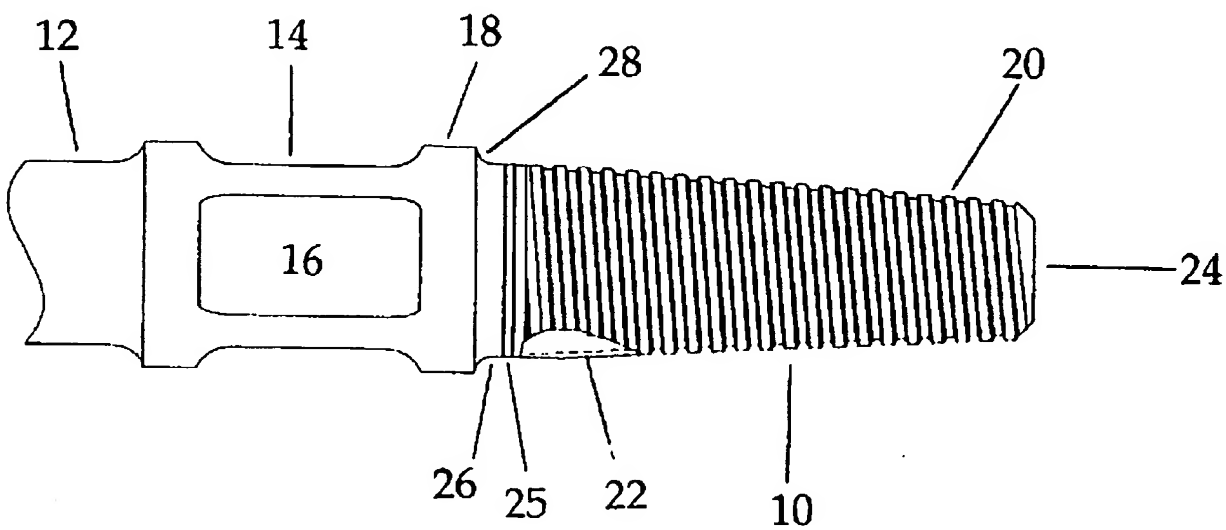

(57) Abrégé/Abstract:
An improved connection and method for making the connection for connecting sucker rods in a sucker rod string used to drive downhole pumps for producing fluid from an underground formation, the connection having an outside diameter no greater than one and one half times the outside diameter of the sucker rod and which uses the contact between the mating threads of the connection for transferring loads between connected sucker rods coupling.

SUCKER ROD COUPLING

Field of the Invention

This invention relates to sucker rods and their connections and in particular to improvements in design of the pin and box connections.

Background of the Invention

Strings of individually coupled sucker rods have been used in oil and gas wells for transmitting mechanical power to artificial lift devices used in the production of oil and gas. Sucker rods generally transfer power by axial load, driving pumps with a reciprocating motion along the well bore (e.g., beam or rod pumps). In recent years, there has been increasing use of sucker rods to drive pumps that operate in a rotary motion (e.g., progressing cavity pumps). This rotary type of pumping transmits power by a torsional load, or torque, along the rods.

Fittings for connecting together a series of sucker rods to reach the downhole pumps in the formation from which fluids are being pumped have long been standardized, and conventional fittings include a uniform diameter thread and a shoulder on which the pin end and box end meet. An example of this type of connection can be found in U.S. Patent 1,671,458 to Wilson. This conventional design limits the length of thread make-up and hence its ability to withstand torsional stress, which is more acute in sucker rods associated with rotary pumps than with reciprocating pumps.

It is conventional to include as the uppermost rod in a sucker rod string a polished rod, which provides a polished surface to accommodate a mechanical dynamic seal, commonly referred to within the industry as the "stuffing box," at the wellhead between the high pressure annulus of the well and the atmosphere. A conventional polished rod has a partially tapered pin/box arrangement with the taper occurring at the end of the threaded section only, and with the taper obtained by reduction of the thread height and not by reduction in diameter of the threaded section as a whole. The purpose of the polished rod taper is to allow the rod to penetrate through the stuffing box without causing damage to the sealing materials contained within. An example of such a conventional tapered pin/box arrangement is shown in U.S. Patent 2,690,934 to Holcombe.

The American Petroleum Institute has set standards for the dimensions of sucker rods and their associated couplings which can be found in API Specification 11B, 25th Edition, January 1, 1995. For sucker rod pin connections, these standards include minimum and maximum threads,

pin-shoulder face parallelism, minimum and maximum diameter of the stress relief section of the pin, minimum and maximum diameter for pin-shoulder and upset bead, minimum and maximum pin length and pin stress relief lengths. Similarly, box connection standards are specified, including nominal thread diameter, total box depth, total thread lengths in the box, including counter bore, minimum box major diameter, maximum box pitch diameter, maximum and minimum box minor diameter and diameter of the box counter bore. Pin and box contact dimensions are similarly specified. Thread forms are specified with reference to the ANSI/ASME B1.1 definition.

All of these dimensional standards are directed to couplings that are intended to transmit power by axial reciprocation. The connection upset reduces the fatigue stress associated with reciprocation and the mating shoulder faces of the box and pin provide a positive make-up indicator and prevent the connection from "breaking out" during operation. High torque capacity is a secondary consideration in the industry-standard design.

In addition to the specific dimensions just listed, external dimensions of couplings and subcouplings are listed for each standard sucker rod size for both conventional couplings and "slim hole" couplings. Whether slimhole or standard, prior art coupling design has included as its largest diameter dimension an upset on the pin end which makes the transition from the shoulder to the base of the wrench flats, which provides the metal mass needed for the torque shoulder to transfer torque through the connection. The diameter of that upset relative to the rod body diameter can be referred to as an upset ratio, which for normal couplings under the API specifications is about two and one eighth to one, and for slimhole couplings under the API specifications is two to one.

As can be appreciated, the space occupied by the coupling within the annulus through which fluids are drawn to the surface diminishes the space available in the annulus for production fluid to flow, resulting in higher friction losses in the fluid. A larger coupling diameter also increases the tubing diameter required for a desired level of production. It would therefore be desirable to decrease the space occupied by the coupling yet maintain the structural integrity needed for the coupling, while in service under axial and torsional load conditions. Eliminating the need for a torque shoulder would significantly reduce the upset ratio thereby providing more annulus space for a given tubing size or permit the use of smaller tubing for effective fluid production.

Reducing the coupling diameter also decreases the standoff between the rod and tubing. This reduces the fatigue weakening common in the rod body adjacent to conventional pin/box connections that are subjected to combined axial and torsional loads in well intervals with moderate to high curvature. It can be appreciated that rods in deviated wells are subjected to cyclic bending stresses as the rod rotates. Furthermore, axial tension on the rod generates a localized curvature concentration adjacent to connections because of the standoff from the tubing wall. By reducing the

connection upset ratio, the standoff is lowered, thereby decreasing the curvature concentration in the adjacent rod, thus improving the fatigue resistance.

Summary of the Invention

The invention includes an improved sucker rod coupling or connection that eliminates the mating pin and box shoulders and provides for torque transfer solely by way of the pin and box mating threads. The invention includes a connection having pin threads formed on a tapered pin body and correspondingly mating threads formed in the bore of the box. In particular, the connection includes a pin having its thread formed on a core that is outwardly tapered from its terminal end to the end of the threaded portion of the pin and a non-threaded section to accommodate a box overhang portion, with the non-threaded portion having an outside diameter approximately the same as the largest diameter of the threaded portion and having a slightly radiused transition between the overhang and a wrench flats section of the pin. With this arrangement, the largest outside diameter of the torque make-up or wrench flats section of the pin can be no greater than the outside diameter of the mated connection. The box portion of the connection is correspondingly shaped and threaded to mate with the pin with load transfer contact between the pin and box provided only by way of the mating threads.

As can be appreciated the coupling can be entirely integral with a rod body, i.e. pin end formed on one rod body end and box on the opposite, or could include a separate box connector having two opposing boxes for mating with rods having pins formed on both rod ends.

The invention further includes a method for optimizing the dimensions and configuration of a connection which eliminates the need for a torque shoulder. The method includes matching the wrench flats section diameter to the sucker rod to be coupled, selecting a thread profile or form and thread length, and selecting bore and core tapers to match the thread length and profile. One feature of the method is selection of a thread form for the connection which, when pin and box portions are engaged, results in contact on both the load and stab flanks of the thread to provide for load transfer between rods to occur in the mated threads.

Brief Description of the Drawing

A better understanding of the present invention can be gleaned from the following detailed description of a preferred embodiment read in light of the accompanying Drawing in which:

Figure 1 is a side view of a sucker rod pin formed in accordance with the instant invention;

Figure 2 is a side view of a sucker rod mated connection formed in accordance with the instant invention;

Figure 3 is a side, cross sectional, schematic view of a box and pin mated connection illustrating the connection geometry;

Figure 4 is a side view showing a preferred thread form for the mated connection of

Figure 2;

Figure 5 is an exploded view showing the detail of a preferred thread form for use in the connection of the invention;

Figure 6 is a graph showing radial load on a coupled rod as a function of the depth of the box counter bore for a one inch rod using a 1.25 inch connection in accordance with the present invention; and

Figure 7 is a graphical comparison of two prior art connections with a connection in accordance with the present invention illustrating improved area available around the connection for production through production tubing.

Detailed Description of a Preferred Embodiment

The basic configuration of the connection according to the invention is shown in Figures 1 and 2 showing the pin and box portions of the connection. With reference to Figure 1, pin 10 is shown formed on the end of rod body 12 and includes a make-up section 14 having wrench flats 16 for assembling and torquing up the connection. Conventionally, the minimum diameter across the centers of the wrench flats matches or is only slightly larger than the outer diameter of the rod body, and corners 18 of the make-up section 14 are sized to provide the notch for holding a wrench in the flats 16.

The pin 10 includes a continuous pin thread 20 formed on tapered core 22. The pin thread 20 and tapered core 22 extend from terminal end 24 of the pin up to a short unthreaded pin connection entrance section 26. The pin entrance section 26 preferably includes a radiused transition 28 to the point where it meets the corners 18 of the make-up section 14. With reference to Figure 2 mated connection 29 is shown having opposing entrances 26', box 30 mated with a pair of pin ends of a sucker rod string. As shown, the box 30 includes an unthreaded entrance 26" and a tapered bore 31 having a continuous thread 32 formed for mating with the pin thread 20. As can be appreciated, the connection can be made either with a rod forming opposing pin ends on the rod body and providing a separate box having opposing boxes for mating with the rod body pin ends or by providing a rod body with one shaped pin end and an opposing end shaped as a box. As can be appreciated, to minimize the chance for well fluid contact with the threads 22 and 32, it may be desirable to include a coupling seal such as seal 25 illustrated in Figure 1. As will be appreciated, any type of sealing mechanism suitable for use with equipment subjected to produced fluids can be used with the coupling of the instant invention. Thread dope should also be used for this purpose as well as for minimizing friction on make-up of the connection, as discussed below.

The connection of the instant invention can be configured to provide the minimum overall connection diameter while providing appropriate strength to transfer the full load capacity, in tension, torsion, or combined loading, of a sucker rod body across the connection. To that end, the connection should be configured for a particular rod body size. It has been found that high torque loads can be transferred through a connection without including a torque shoulder in the connection using a tapered pin core and box bore and using only the threaded interface for load transfer by choosing an appropriate taper and thread length for the connection. It is also advantageous to provide a thread geometry which maximizes contact between the pin and box throughout the thread length.

Figure 4 schematically illustrates connection geometry and the manner in which a tapered thread core is used to produce radial interference when the pin 10 is advanced in the box 30. One key feature of the invention is the inclusion of box overhang 34 outside of the threaded engaged length of the coupling. This overhang 34 produces a radial force concentration at the mated connection entrance section 26'. Because the box wall is thinnest in the mated connection entrance section 26', the lowest inward radial forces are seen in this section, some of which are transferred from the overhang 34. It has been found that an optimal length of the overhang 34 can be determined using bending wave equations as described below.

Although any thread design can be used, there are some thread types that are known to be more effective at transferring torque. For most effective torque transfer, the threads 20 and 32 preferably include straight flanks, e.g., load flanks 36 and stab flanks 38 and a flat root/crest, e.g., pin crest 40 and box root 42 as best seen in Figure 5. In general, the thread height should be kept small to minimize the effect of the thread on coupling wall thickness.

Turning now to the method for optimizing the physical parameters of the coupling, the following will first address the forces on the critical sections of the coupling as secured to a rod body, including the rod body 12, the pin connection entrance 26, the coupling make-up or mid-section 14 and its thread section including the threads 20 and 32. Load transfer equations are then applied to the results of the critical section designs. A key feature of the design optimization method of the invention is to use as a model for the coupling load a thick-wall-pressure-vessel. It has been found that, although a coupling is not a pressure-vessel, mathematical models developed for stresses in such a vessel lead to design results that are effective to produce a coupling capable of effective torque transfer and of sufficient structural integrity to withstand the rigorous forces to which a sucker rod connection is subject in use.

Critical Sections

Rod Body 12

The rod body section is a circular section. Hollow rods have an opening down the centre of the rod. The rod body section capacities are given in terms of the rod diameter and rod bore diameter as follows:

$$F_y = F_u = \sigma_y \frac{\pi(d_r^2 - d_i^2)}{4} \quad \text{(axial load capacity)}$$

$$T_y = \frac{\sigma_y}{\sqrt{3}} \frac{\pi(d_r^3 - d_i^3)}{16} \quad \text{(torsional yield capacity)}$$

$$T_u = \frac{\sigma_y}{\sqrt{3}} \frac{\pi(d_r^3 - d_i^3)}{12} \quad \text{(torsional ultimate capacity)}$$

where:

σ_y = tensile material strength,

d_r = rod diameter, and

d_i = rod bore diameter (hollow rods).

In the following, only solid rods will be considered for simplicity. However, as will be appreciated the optimization method can also be applied to hollow rods. Since hollow rods have lower section load capacities, connection designs meeting load requirements for solid rods will more than meet loads for hollow rods.

Mated Connection Entrance 26'

Load is transferred across the threads from the pin to the box along the taper of the threaded zone. Torsional load is transferred by friction, while axial load is transferred mechanically by bearing loads on the thread flanks. If the load transfer rate with respect to location is insufficient, the section capacity is reduced by the taper faster than the load is transferred out of the pin, which would lead to a failure in the pin.

Given the taper, t, of the thread in terms of the initial and final engaged thread diameters (d_o and d_e respectively) and the engaged thread length (L_t) as shown in Figure 3,

$$t = \frac{d_o - d_e}{L_t}$$

where the thread pitch diameter, d, at any location, z shown exploded in Figure 5, can be expressed as

$$d = d_o - tz$$

The ultimate section capacity gradients can be shown to be:

$$\frac{dF_u}{dz} = -\sigma_y \frac{\pi t(d_o - tz)}{2}$$

$$\frac{dT_u}{dz} = -\frac{\sigma_y}{\sqrt{3}} \frac{\pi t(d_o - tz)^2}{4}$$

These relationships show that the largest section capacity gradient occurs where the thread diameter is largest and the box thickness is smallest, at the mated connection entrance 26'.

Coupling Make-Up Section or Mid-Section 14

With continuing reference to Figures 3, 4, and 5, the mid-section 14 of the coupling carries the full rod body 12 loads between pins. In matching the coupling mid-section 14 capacity to the rod body 12, only the ultimate capacities need be considered. The section limits are expressed similarly to those for the rod body 12:

$$F_u = \sigma_y \frac{\pi(d_c^2 - d_e^2)}{4} \quad \text{(tensile limit)}$$

$$T_u = \frac{\sigma_y}{\sqrt{3}} \frac{\pi(d_c^3 - d_e^3)}{12} \quad \text{(torsional limit)}$$

where d_c is the outside diameter of the box.

For a given box diameter, d_c, the maximum end diameter to match the rod body 12 capacities can be determined, assuming similar material strengths for the rod and box:

$$d_{e(axial)} = \sqrt{d_c^2 - d_r^2}$$

$$d_{e(torsional)} = \sqrt[3]{d_c^3 - d_r^3}$$

The box diameter is expressed in terms of the nominal rod diameter and an upset parameter, β:

$$d_c = \beta d_r$$

$$d_{e(torsional)} = d_r \sqrt[3]{\beta^3 - 1}$$

$$d_{e(axial)} = d_r \sqrt{\beta^2 - 1}$$

Equation 1

The smallest box inside diameter must be used to ensure the section strength is adequate over the range of combined load conditions that may be encountered. Consequently, the axial load criterion governs the inside diameter of the coupling. If manufacturing constraints impose a maximum coupling inside diameter, then this relationship can be used to define the coupling upset required.

Thread Sections 20 and 32

Loads are transferred across the contact surfaces on the threads and into the bodies of the pin and box through the base of the threads. In most cases the frictional characteristics

require a sufficiently long threaded section that the thread limits are not of concern. However, for heavier upset connections, the thread strength can govern the design.

Figure 4 shows the critical section for one thread. The thread width over which the load is transferred is a fraction of the total thread pitch. Therefore, the stress transferred across the critical thread sections 20 and 32 can be expressed in terms of the average load transfer:

$$\sigma_t = \sigma_{avg}\frac{P}{w} = \frac{\sigma_{avg}}{\eta} \qquad \text{Equation 2}$$

$$\tau_t = \frac{\tau_{avg}}{\eta}$$

where w is the thread width at the critical location and P is the thread pitch. The thread efficiency factor η indicates what proportion of the thread cone carries the load. For fully engaged V-threads, the thread efficiency approaches 100%. For square threads, the thread efficiency is roughly 50%, and for partially engaged V-threads the thread efficiency can be 25% to 50%. Assuming a 50% thread efficiency is slightly conservative for the thread type preferred for this application.

The thread sections 20 and 32 will fail when the stress state on the entire thread sections 20 and 32 (i.e. on all threads) reaches the yield limit:

$$\sigma_{avg} = \frac{F}{A} = \frac{2F}{\pi(d_o + d_e)L_t}$$

$$\tau_{avg} = \frac{T}{\int_{z=0}^{L_t} \pi\frac{d^2}{2}dz} = \frac{6Tt}{\pi\left[d_o^3 - (d_o - tL_t)^3\right]} = \frac{6T}{\pi\left[d_o^2 + d_o d_e + d_e^2\right]L_T}$$

<u>Coupling Hoop Limit</u>

The coupling is expanded by radial interference as the pin is advanced into the box, developing the radial stress required to produce the circumferential friction force. The radial force that can be developed is limited by the strength of the coupling material, and by the thickness of the coupling. If the friction factor is insufficient, or the length of the connection is too short, the radial force required to produce the necessary torque may exceed the capacity of the coupling. This leads to failure of the coupling by hoop expansion, perhaps to the point where the coupling splits.

<u>Load Transfer Equations</u>

<u>Frictional Torque</u>

Torque is developed by friction produced by the radial load resulting from radial interference. The coupling thickness is significant relative to the coupling diameter, so thick wall pressure vessel equations are appropriate to relate the radial force to the interference. In

this development the radial interference will be assumed constant over the length of the threads. It is a simple matter to extend the design criteria to account for a linear interference distribution associated with a taper mismatch between the pin and box.

Figure 5 shows a free body diagram of an infinitesimal interval of the coupling subject to radial interference. The thick wall pressure vessel equations for an uncapped vessel can be expressed giving the contact stress C in terms of the diametrical interference I (twice the radial interference), geometric characteristics, and elastic material properties.

$$C = \frac{IE\left(d_c^2 - d^2\right)}{d\left[\left(1+\nu\right) d_c^2 + \left(1-2\nu\right) d^2\right]},$$

where E is the elastic modulus and ν is Poisson's ratio.

The coupling inside diameter (or thread diameter) is expressed in terms of the coupling outside diameter by a factor α, and the contact stress is updated correspondingly:

$$d = \alpha d_c$$

$$C = \frac{IE\left(1-\alpha^2\right)}{\alpha d_c\left[(1+\nu)+\alpha^2(1-2\nu)\right]}$$

The friction torque associated by this contact stress on the infinitesimal interval depends on the effective frictional characteristic μ_e:

$$dT = \frac{\pi \mu_e I E d_c}{2} \frac{\alpha\left(1-\alpha^2\right)}{\left[(1+\nu)+\alpha^2\left(1-2\nu\right)\right]} dz$$

$$\mu_e = \frac{\mu}{\sin(\phi/2)}$$

Equation 3

This differential equation is the basis for two of the most important design equations for the connection. First, the maximum allowable thread taper can be calculated to prevent a failure in the pin connection entrance section 26 of the mated connection 29. From this calculation the threaded length can be determined and the total torque transferred can be calculated from the integration of the differential equation.

The torque transfer rate must be greater than the section torque capacity gradient at the mated connection entrance 26":

$$\frac{dT_f}{dz} \geq -\frac{dT_u}{dz}$$

$$\frac{\pi \mu I E d_c}{2} \frac{\alpha\left(1-\alpha^2\right)}{\left[\left(1+\nu\right)+\alpha^2\left(1-2\nu\right)\right]} \geq \frac{\sigma_y}{\sqrt{3}} \frac{\pi t\left(d_o - tz\right)^2}{4}.$$

$$t \leq \frac{2\sqrt{3}\mu I E}{\sigma_y \alpha d_c} \frac{(1-\alpha^2)}{[(1+\nu) + \alpha^2(1-2\nu)]}$$

Equation 4

The total frictional torque transferred across the threads is evaluated from integrating the differential equation. First, the ratio α_o of the thread pitch diameter d_o to the coupling diameter d_c is expressed in terms of the axial position z:

$$\alpha = \alpha_o - mz, \quad \text{where } m = \frac{\alpha_o - \alpha_e}{L}$$

$$dz = \frac{-1}{m} d\alpha$$

$$T_f = \int_{\alpha=\alpha_o}^{\alpha_e} \frac{\pi \mu_e I E d_c}{-2m} \frac{\alpha(1-\alpha^2)}{(1+\nu) + \alpha^2(1-2\nu)} d\alpha$$

The following integration formula can be used with appropriate variable substitutions to integrate the equation:

$$\int \frac{u(1-u^2)}{a+bu^2} du = \frac{a+b}{2b^2} \ln(a+bu^2) - \frac{u^2}{2b}$$

$$a = (1+\nu), b = (1-2\nu), u = \alpha$$

The total frictional torque is thus:

$$T_f = \frac{\pi \mu_e I E d_c L}{4(\alpha_o - \alpha_e)} \left[\frac{2-\nu}{(1-2\nu)^2} \ln\left(\frac{(1+\nu) + (1-2\nu)\alpha_o^2}{(1+\nu) + (1-2\nu)\alpha_e^2} \right) - \frac{\alpha_o^2 - \alpha_e^2}{1-2\nu} \right]$$

Equation 5

This expression is valid as long as the coupling remains elastic. The effective stress in the coupling is largest on the coupling inside diameter at the end of the thread, adjacent to the coupling mid-section 14. The elastic torque capacity of the thread is reached when the interference produces a von Mises effective stress at this location. The amount of interference at the yield limit is determined using the thick wall pressure vessel equations:

$$C_y = \frac{\sigma_y(1-\alpha_e^2)}{\sqrt{3+\alpha_e^2}}$$ (contact stress at first yield)

$$I = C_y d_c \frac{(1+\nu) + (1-2\nu)\alpha_e^2}{E(1-\alpha_e^2)}$$

$$I = d_c \frac{\sigma_y}{E} \frac{[(1+\nu) + (1-2\nu)\alpha_e^2]}{\sqrt{3+\alpha_e^2}}$$

Equation 6

The interference associated with first yield of the coupling is used to determine a coupling geometry that can transfer the ultimate rod body 12 torque. The additional plastic

capacity of the connection accounts for the multidimensional stress effects resulting from the rod body 12 loads that are transferred through the coupling mid-section 14 simultaneously with the interference loads.

Axial Force

The mechanical load transfer rate is governed by the shear capacity of the critical thread sections 20 and 32 of the mated connection 29:

$$\frac{dF_m}{dz} = \pi d\eta \frac{\sigma_y}{\sqrt{3}}$$

Axial load transfer requirements at the mated connection entrance 26' are similar to those for torque:

$$\frac{dF_m}{dz} \geq -\frac{dF_u}{dz}$$

$$\pi d\eta \frac{\sigma_y}{\sqrt{3}} \geq \sigma_y \frac{\pi t(d_o - tz)}{2}$$

$$t \leq \frac{2\eta}{\sqrt{3}}$$

Equation 7

In practical applications the maximum taper allowed by the axial load transfer criterion is much larger than that allowed by the torque transfer criterion.

Thread Load Limits

Thread load limits are calculated based on the assumption that the stress transferred across the critical thread sections 20 and 32 reaches the material yield limit over the entire threaded region. A thread capacity safety factor is determined from the quotient between the thread load capacity and the rod body 12 capacity. Since the threads are also subjected to large bearing forces on the thread flank, it is recommended that at thread capacity safety factor of at least two be maintained in the design. For most minimal upset designs frictional torque transfer rate considerations govern, producing thread capacity safety factors significantly higher than two. It is possible for the thread capacity to govern when the coupling upset becomes more significant.

For design, the calculations for thread capacity is based on the area of the cone defined by the thread roots. This introduces a slight conservatism in the design because this cone diameter is slightly smaller than that of the critical thread area. The difference between the thread pitch diameter and the root diameter is equal to the thread height, h.

Torsional Thread Capacity

The ultimate torsional load capacity, T_T, of the critical thread sections 20 and 32 under pure shear is given by the following integration:

$$T_T = \frac{\pi\eta\sigma_y}{2\sqrt{3}} \int_{z=0}^{L} (d-h)^2 dz$$

$$d = d_o - tz, \quad t = \frac{d_o - d_e}{L} = \frac{(d_o - h) - (d_e - h)}{L}$$

$$T_T = \frac{\pi\eta\sigma_y}{2\sqrt{3}} \int (d_o - h - tz)^2 dz$$

$$T_T = \frac{\pi\eta\sigma_y}{2\sqrt{3}} \left. \frac{[d_o - h - tz]^3}{-3t} \right|_0^L$$

$$T_T = \frac{\pi\eta\sigma_y}{6\sqrt{3}t} \left[(d_o - h)^3 - (d_e - h)^3 \right]$$

$$T_T = \frac{\pi\eta\sigma_y L}{6\sqrt{3}} \left[(d_o - h)^2 + (d_o - h)(d_e - h) + (d_e - h)^2 \right]$$

Equation 8

where,

d_o = thread diameter at coupling entrance 26,

d_e = thread diameter in centre of coupling,

h = thread height

η = thread efficiency factor

Axial Thread Capacity

The ultimate axial load capacity F_T of the critical thread sections 20 and 32 under pure shear is given by the following integration, using similar substitutions to those used in the torsional thread capacity:

$$F_T = \frac{\pi\eta\sigma_y}{\sqrt{3}} \int_{z=0}^{L} (d-h) dz$$

$$F_T = \frac{\pi\eta\sigma_y}{\sqrt{3}} \int_{z=0}^{L} (d_o - h - tz) dz$$

$$F_T = \frac{\pi\eta\sigma_y L}{\sqrt{3}} (d_o + d_e - 2h)$$

Equation 9

Counterbore Optimisation

If the connection entrance or coupling mouth extends past the thread interference zone, e.g., includes unthreaded section 44 shown in Figure 3, elastic deformation energy generates an additional radial force in the first threads. This radial force is modest because the D/t ratio is largest at that location, where D is one half of the average of the coupling outside and inside diameters, and t is one half of the difference between the coupling outside and inside diameters. The additional radial force at the mated connection entrance 26' can augment the initial torque

transfer and reduce the threaded length required to transfer torque, or provide a modest safety factor in the torque transfer mechanism. The box wall thickness in the counterbore is relatively small in comparison with the connection diameter, so the equations for a beam on an elastic foundation can be used to estimate the ring force produced by the overhang 34.

The spring stiffness, k, and wavelength parameter, β, are given as:

$$k = \frac{4Et}{d^2}$$

$$\beta = \sqrt[4]{\frac{12(1-\nu^2)}{d^2 t^2}}$$

The equations for maximum displacement and moment in a short beam on an elastic foundation are:

$$u_{max} = \frac{P\beta}{2k} \frac{\cosh \beta L + \cos \beta L + 2}{\sinh \beta L + \sin \beta L}$$

$$M_{max} = \frac{P}{4\beta} \frac{\cosh \beta L - \cos \beta L}{\sinh \beta L + \sin \beta L}$$

The length of the short beam, L, is twice the overhang length for the coupling L_c, and P is also twice the augmentation load P_c.

The displacement is equal to half of the diametrical interference I. Solving for the augmentation load P and graphing with respect to the counterbore length (Figure 6 for a 1" connection) illustrates that the primary benefit is developed within 0.25 inches, which corresponds to one half of the characteristic wavelength.

When the counterbore length has been finalised the torque associated with the counterbore augmentation is calculated by:

$$T_a = \frac{\pi \alpha_o^2 P_c d_c^2}{2} \qquad \text{Equation 10}$$

Using the above equations and parameters, optimal dimensions of a coupling in accordance with the present invention that is suitable for use in the field can be determined as follows:

<u>Design Optimization Method</u>

Coupling diameter selection: The coupling mid-section area is matched to the rod area using Equation 1. For designs constrained by manufacturing limitations on the inside diameter of the box, the coupling upset is defined by the rod diameter and minimum allowable (by manufacturing limits) coupling inside diameter. If a larger diameter coupling is required to facilitate handling procedures, the coupling inside diameter is defined by the rod and coupling diameters. Maximizing the coupling inside diameter increases the torque transferred over a

given thread length, so there is no advantage to reducing the inside diameter further than necessary for a given coupling upset.

Calculate coupling yield interference: Using the selected coupling diameter, the coupling yield interference is determined using Equation 6.

Thread profile selection: The thread profile should then be selected to define the effective friction coefficient for the frictional torque load transfer calculation, and the thread efficiency parameter for the critical thread section load capacities. The effective friction coefficient is given by Equation 3, and the thread efficiency is defined in Equation 2. The thread height must be reduced as much as possible to minimize the impact on the coupling hoop strength and stiffness at the mated connection entrance 26'. Thread dope should be selected to give the minimum friction coefficient at make-up. This ensures that the torque capacity will not degrade if well bore fluids migrate into the threads over time. Connections should be made-up to the specified torque limit.

Thread length calculations for load transfer in threads: Four load transfer calculations are performed to calculate the minimum threaded length required to transfer load for all critical sections. Typically, the torque transfer criteria govern. Threaded lengths required for torque transfer are calculated using Equations 4 and 5. The threaded length based on an axial load transfer criterion is calculated using Equation 7. The longest thread length evaluated from these calculations is used in the subsequent step.

Threaded section capacities check: The thread capacities are checked by evaluating the safety factors, using the threaded length determined in the previous step. Equations 8 and 9 give the ultimate thread capacity of the connection. Dividing these results by the respective rod body ultimate capacities gives the safety factors with respect to thread section limits. If the safety factor is less than that desired, the thread length should be scaled up in proportion to the deficiency.

Coupling counterbore calculation: The counterbore torque is calculated by Equation 10. This torque, or a portion of it, can be used to provide an additional safety factor. If the thread length is governed by the frictional torque transfer criterion, the portion of this value that isn't used for a safety margin can be used to reduce the threaded length further, as described in the following step.

Revise thread length for designs governed by friction: If frictional torque transfer governs the threaded length, the counterbore torque load can be used to further optimise the coupling size.

To illustrate the use of the method of the present invention, the following Sample Design Calculation is presented. As can be appreciated, the invention is not limited to the particular

dimensions resulting from any calculation of optimal design, but rather the scope of the invention is defined by the scope of the claims at the end of this description. The sample is presented merely to summarize and illustrate one method for optimizing the design of a coupling for a particular common sucker rod diameter in accordance with the invention.

<u>Sample Design Calculation</u>

A sample design is presented to demonstrate one optimisation approach using the new design equations. In this example, a connection design is developed for a 1 inch solid rod with a 0.25 inch upset connection on the diameter. A material strength of 100,000 psi is assumed for the exercise, and the common values for elastic modulus and Poisson's ratio are used: 30×10^6 and 0.3, respectively. The following does not include design considerations for tolerancing or manufacturing.

<u>Step 1:</u> Calculate the bore diameter using Equation 1. The upset ratio is calculated from the coupling and rod diameters and the maximum allowable coupling bore diameter is determined using the axial section capacity:

$$\beta = \frac{d_c}{d_r} = 1.25$$

$$d_{e(axial)} = d_r\sqrt{\beta^2 - 1} = 1\sqrt{1.25^2 - 1} = 0.562$$

<u>Step 2:</u> Calculate the thread interference (on the diameter) to produce first yield in the coupling using Equation 6:

$$\alpha_e = \frac{d_e}{d_c} = \frac{0.5626}{1.25} = 0.45$$

$$I = d_c\frac{\sigma_y}{E}\frac{\left[(1+\upsilon)+(1-2\upsilon)\alpha_e^2\right]}{\sqrt{3+\alpha_e^2}}$$

$$I = d_c\frac{\sigma_y}{E}\frac{\left[(1+\upsilon)+(1-2\upsilon)\alpha_e^2\right]}{\sqrt{3+\alpha_e^2}}$$

$$I = 1.25\frac{100{,}000}{30\times10^6}\frac{\left[(1+0.3)+(1-2\times0.3)0.45^2\right]}{\sqrt{3+0.45^2}}$$

$$I = 0.003215$$

<u>Step 3:</u> Define the basic thread form characteristics so that an effective friction factor and thread efficiency may be defined. For this exercise a thread height of 0.025 is defined with symmetric load and stab flanks at 22.5° from the plane perpendicular to the rod axis for an included angle, ϕ, of 45° between the two flanks. A symmetric thread giving equal thread width to the pin and box is used to optimise the thread efficiency. In the example, the threads

are assumed to mate perfectly and transfer load only on the thread flanks. A coarse thread pitch of 4 TPI (threads per inch) is used.

A conservative friction coefficient of 0.1 is assumed. The effective friction factor is calculated using Equation 3:

$$\mu_e = \frac{\mu}{\sin(\phi/2)} = \frac{0.1}{\sin(22.5^\circ)} = 0.26$$

The thread efficiency is the ratio of the thread shear area to the total thread area. The total thread width at the pitch line is 0.125 inches, and the width at the base of the thread is:

$$w = 0.125 + 2\tan(22.5^\circ)(0.025/2) = 0.135$$

$$\eta = \frac{w}{P} = \frac{0.135}{0.250} = 0.54$$

Step 4: Calculate the engaged thread length required for various critical section criteria. These equations are defined in terms of the thread pitch diameters. Therefore, the diameters used in these calculations reflect the pitch diameter of the thread.

The torque transfer rate criteria defining the maximum allowable thread taper is given by Equation 4. The calculation is made for the mated connection entrance 26' of the coupling where the torque transfer requirement is most demanding.

$$t \leq \frac{2\sqrt{3}\mu_e IE}{\sigma_y \alpha d_c} \frac{\left(1-\alpha^2\right)}{\left[(1+\nu)+\alpha^2(1-2\nu)\right]}$$

$$\alpha = \alpha_o = \frac{d_o + h}{d_c} = \frac{1.0+0.025}{1.25} = 0.82$$

$$t \leq \frac{2\sqrt{3}(0.26)(0.003215)(30\times10^6)}{(100{,}000)(0.82)(1.25)} \frac{\left(1-0.82^2\right)}{\left[(1+0.3)+0.82^2(1-2(0.3))\right]}$$

$$t \leq 0.178$$

The engaged thread length is calculated from the taper equation:

$$t = \frac{d_o - d_e}{L_t}$$

$$L_t = \frac{d_o - d_e}{t} = \frac{1.0-.5625}{0.178} = 2.45\,\text{inches} \qquad \textbf{(a)}$$

The axial load transfer rate criteria gives a maximum taper by Equation 7. This would not normally govern the design, but is included here for completeness. The maximum taper allowed by the axial load transfer rate criterion is given by:

$$t \leq \frac{2\eta}{\sqrt{3}}$$

$$t \leq \frac{2(0.54)}{\sqrt{3}} = 0.62$$ **(b)**

This value is over three times larger than that based on the torsional transfer rate criterion and therefore does not govern the design.

The total torsional load transfer capacity is given by Equation 5:

$$T_f = \frac{\pi\mu_t I E d_t L}{4(\alpha_o - \alpha_t)}\left[\frac{2-\nu}{(1-2\nu)^2}\ln\left(\frac{(1+\nu)+(1-2\nu)\alpha_o^2}{(1+\nu)+(1-2\nu)\alpha_t^2}\right) - \frac{\alpha_o^2-\alpha_t^2}{1-2\nu}\right]$$

The ultimate torque capacity of the rod body is given by:

$$T_u = \frac{\sigma_y}{\sqrt{3}}\frac{\pi d_r^3}{12} = \frac{100{,}000}{\sqrt{3}}\frac{\pi(1.0)^3}{12} = 15{,}100\,\text{in.-lbs.}$$

The value for α_t is also updated to the thread pitch line, increasing its value to 0.47 from 0.45. Solving for L so that the torque transfer capacity equals the ultimate rod body capacity gives:

$L = 1.35$ inches **(c)**

The length based on the torque transfer rate is higher than that based on the total torque transfer. Because the engaged length calculated from the taper equation of 2.45 inches is longer (L_t above), it is used in the remaining design steps because it results in the other criteria being satisfied.

<u>Step 5:</u> Check load safety factors on the thread shear area using Equations 8 and 9. The torsional load capacity for the thread shear area is (Equation 8) in terms of the thread pitch line geometry is:

$$T_T = \frac{\pi\eta\sigma_y L}{6\sqrt{3}}\left[(d_o-h)^2 + (d_o-h)(d_t-h) + (d_t-h)^2\right]$$

$$T_T = \frac{\pi(0.54)(100{,}000)(2.45)}{6\sqrt{3}}\left(1.0^2 + (1.0)(0.5625) + (0.5625^2)\right)$$

$$T_t = 75{,}300\,\text{in.-lbs.}$$

$$\text{Torque Saftey Factor} = \frac{75{,}300}{15{,}100} = 4.6$$

The axial load capacity of the thread is compared with the ultimate rod body axial load capacity. The rod body capacity is:

$$F_u = \frac{\pi\sigma_y d_o^2}{4} = \frac{\pi(100{,}000)(1.0)^2}{4} = 78{,}500\,\text{lbs.}$$

The thread capacity is given by Equation 9:

$$F_T = \frac{\pi\eta\sigma_y L}{\sqrt{3}}(d_o + d_e - 2h) = \frac{\pi(0.54)(100{,}000)(2.45)}{\sqrt{3}}(1.025 + 0.5875 - 0.05)$$

$$F_T = 376{,}000 \text{lbs}$$

$$\text{AxialSafetyFactor} = \frac{376{,}000}{78{,}500} = 4.79$$

Both axial and torsional safety factors indicate adequate thread shear area capacity, so the engaged thread length remains at 2.45 inches.

Step 6: The counterbore torque is calculated from Equation 10. Most of the augmentation load is developed with a length of 0.25 inches, at which P_c is ~~463~~ 776 lbs./in. Equation 10 gives a counterbore torque as:

$$T_a = \frac{\pi\alpha_o^2 P_c d_c^2}{2} = \frac{\pi(0.82)^2 \overset{776}{\cancel{(463)}}(1.25)^2}{2}$$

$$T_a = 1280 \text{ in.-lbs.}$$

In this example, the effective counterbore length and the augmentation torque are small because of the wall thickness in the counterbore. An actual design would probably accept this as a modest (8%) improvement in the overall safety factor.

Step 7: Optimisation with counterbore could be done to reduce the overall thread length. The augmentation torque can be used to reduce the effective connection entrance diameter in the thread taper and thread length calculations in Steps 4a and 4c. Assuming the augmentation torque is transferred at the first engaged thread, the remaining torque is:

$$T_{u1} = T_u - T_a = 15{,}100 - 1{,}280 = 13{,}800 \text{in.-lbs}$$

The critical diameter for carrying this torque is calculated from the ultimate torque equation:

$$T_u = \frac{\sigma_y}{\sqrt{3}} \frac{\pi d_r^3}{12}$$

$$d_{o1} = \sqrt[3]{\frac{12\sqrt{3}T_{u1}}{\pi\sigma_y}} = \sqrt[3]{\frac{12\sqrt{3}(13{,}800)}{\pi(100{,}000)}} = 0.97$$

Step 4 can be re-evaluated with d_{o1} replacing d_o. Adjusting the diameters to thread pitch diameters gives:

$$\alpha_{o1} = \frac{d_{o1} + h}{dc} = \frac{0.971 + 0.025}{1.25} = 0.797$$

The thread lengths determined in Steps 4 then become:

L_t = 1.98 in. (compared with 2.45 in.)

L_t = 1.12 in. (compared with 1.25 in.)

However, this is the length from the effective diameter (0.973 inches), not from the thread start. The criteria from step 4a governs, so using the thread taper from that calculation gives the total engaged thread length required:

$$L_t = L_{t1} + \frac{(d_o - d_{o1})}{t_1},\ t_1 = 0.16$$

$$L_t = 1.979 + \frac{(1.0 - .971)}{0.21} = 2.12 \text{inches}$$

In this example the thread length can be reduced by 13% if the counterbore optimisation is used. For thicker wall counterbores the improvement can be even greater, provided the torque transfer rate (Step 4a), or the total torque criteria (Step 4b) governs.

The final sample connection dimensions are therefore as follows:

Rod diameter (d_r)	=	1.00 in.	25.4mm
Coupling diameter (d_c)	=	1.25 in.	31.75mm
Coupling bore diam. (d_o-h)	=	0.5625 in.	14.29mm
Counterbore diam. (d_o+h)	=	1.05 in.	26.67mm
Counterbore length (L_c)	=	0.25 in.	6.35mm
Thread height (h)	=	0.025 in	0.64mm
Flank angle (Φ)	=	22.5° (45° included angle)	
Pitch diameters			
coupling entrance(d_o)	=	1.025 in.	26.04mm
coupling centre (d_o)	=	0.5875 in.	14.92mm
Thread length (L)	=	2.12 in.	53.85mm
Thread pitch (defined term)	=	4 threads per inch (0.25 in./thread) (6.35 mm/threaad)	

Figure 7 illustrates the beneficial end result of the connection of the instant invention over prior art couplings which include a torque shoulder, where Figure 7A shows a standard 1 inch coupling within a typical 2.875 inch diameter tubing, Figure 7B a one inch conventional slimhole coupling and Figure 7C a coupling in accordance with the present invention. As illustrated, a connection designed using the above optimization method for a nominal one inch sucker rod would have a maximum outside diameter of 1.25 inches (3.175 cm) and a length of 5.8 inches (14.73 cm) and will provide as much as 263 percent more flow area about the coupling than prior art conventional couplings designed for the same rod body 12. Furthermore, with respect to fatigue stresses, the reduction in standoff reduces fatigue stresses appreciably. For example, for the 1" rod in the sample design under a 15,000 lb. tensile load in

a well section with a curvature of 15 degree/100ft, the bending concentration is only 2.3 times the nominal curvature or 53% the bending concentration of 4.3 produced by conventional couplings.

As can now be appreciated, the invention is not limited to the parameters and examples which are given above for the purpose of teaching how to practice the invention, but rather is defined by the following claims.

What is claimed is:

1. An improved connection for use in connecting sucker rods having an outside diameter and a full load capacity in a sucker rod string used to drive downhole pumps that pump fluids from an underground formation to the surface including a pin formed on an end of the sucker rod to be coupled, wherein the pin includes a tapered, threaded portion for coupling with a box having a correspondingly tapered, threaded bore to provide a box and pin connection having a taper, a thread length and a connection load capacity, the improvement comprising sizing the box and pin connection to have a maximum outside diameter when mated that is no greater than about one and one half times the outside diameter of the sucker rod to be coupled and wherein the tapered, threaded portion of the pin has opposing ends defining the tapered, threaded portion therebetween and is comprised of a core having threads formed thereon, wherein the core has a core diameter and a core length and wherein the taper is defined as the difference between the core diameters at opposing ends of the tapered, threaded portion divided by the core length between the opposing ends of the tapered, threaded portion and the taper is between .1 and .44 such that the connection load capacity meets or exceeds the full load capacity, including torsional load, of the sucker rod across the connection.

2. The improved connection of claim 1 wherein the connection further includes a make-up section and wherein the make-up section has its maximum nominal outside diameter no larger than the maximum outside diameter of the box and pin connection when mated.

3. The improved connection of claim 1 wherein the box is comprised of a separate box having an internal bore inwardly tapering from opposing ends and having threads formed on the bore which are configured to mate with the threaded portions of the pins of successive rods to be coupled.

4. The improved connection of claim 1 wherein the ratio of the thread length to the diameter of the sucker rod to be coupled is between .9 and 6.5.

5. The improved connection of claim 1 wherein the preselected thread form includes a flat pin crest and opposingly sloping load and stab flanks.

6. A connection for connecting sucker rods in a sucker rod string used to pump fluids from an underground formation to the surface comprising:

a pin formed on an end of a sucker rod to be coupled having threads formed on a tapered core to provide a threaded portion of the core, wherein the threaded portion of the core of the pin has a terminal end and an opposite end and wherein the threaded portion of the core of the pin is outwardly tapered from its terminal end, and wherein the core has a core diameter and the threaded portion of the core defines an axial length between the terminal end and the opposite end of the threaded portion, and the outward taper of the core is defined by the ratio of the core diameter at its terminal end minus the core diameter at its opposite end to the axial length of the threaded portion and said ratio is between .1 and .44; and

a box having a correspondingly tapered bore therein with mating threads on the bore for coupling with the pin to provide a box and pin connection in which all coupling load in use is born by the mating threads.

7. The connection of claim 6 wherein the ratio of the thread length to the diameter of the sucker rod to be coupled is between .9 and 6.5.

8. A method for optimizing the dimensions of a sucker rod connection comprised of a connection material and having an outside diameter for use in connecting sucker rods in a sucker rod string used to drive downhole pumping mechanisms to pump fluids from an underground formation to the surface comprising:

Providing a threaded box having an internal, tapered bore, the bore defining a bore taper, and a correspondingly tapered pin formed on an end of a sucker rod to be coupled, the pin having corresponding threads formed on a core of the pin for coupling with the threaded box, the core defining a corresponding core taper, wherein the corresponding pin and box threads have a thread length which satisfies the equation

$$L \geq \frac{4(\alpha_o - \alpha_e)\alpha_o^3 \sigma_y d_c^2}{12\sqrt{3}\mu_e IE\left[\frac{2-\nu}{(1-2\nu)^2}\ln\left(\frac{(1+\nu)+(1-2\nu)\alpha_o^2}{(1+\nu)+(1-2\nu)\alpha_e^2}\right) - \frac{\alpha_o^2 - \alpha_e^2}{1-2\nu}\right]}$$

and wherein the corresponding core and bore tapers satisfy the equation

$$t \leq \frac{2\sqrt{3}u_e(1-\alpha_o^2)}{\alpha_o\sqrt{3+\alpha_e^2}} \frac{[(1+\nu)+\alpha_e^2(1-2\nu)]}{[(1+\nu)+\alpha_o^2(1-2\nu)]}$$

where:

d_C is the outside diameter of the connection;

α_e is the ratio of the thread diameter to the outside diameter of the connection at a terminal end of the pin thread;

α_o is the ratio of the thread diameter to the outside diameter of the connection at a terminal end of the box thread;

E is the Young's modulus material property for the connection material;

$\mu_e = \dfrac{\mu}{\sin(\phi/2)}$ is an effective friction coefficient, where μ is the nominal friction coefficient of the thread interface, and ϕ is the thread flank angle;

σ_y is the yield strength of the connection material;

ν is Poisson's ratio material property for the connection material; and

I is the coupling yield interference defined by the equation:

$$I = \frac{d_c\sigma_y[(1+\nu)+(1-2\nu)\alpha_e^2]}{E\sqrt{3+\alpha_c^2}}$$

9. A method for making a sucker rod coupling having a coupling outside diameter and including a threaded pin portion and correspondingly threaded box portion for use in

connecting sucker rods having a rod outside diameter in a sucker rod string used to drive downhole pumping mechanisms to pump fluids from an underground formation to the surface comprising the steps of:

(a) selecting a coupling outside diameter no greater than about one and one half times the rod outside diameter of the sucker rod to be coupled and a coupling inside diameter no greater than the square root of the coupling outside diameter squared minus the square of the rod outside diameter;

(b) determining the coupling yield interference for the selected coupling using a thick wall pressure vessel as the mathematical model for the calculated interference to determine radial and tangential stresses for the coupling and thereby determining the torque required for the coupling when the pin portion and box portion are mated; and

(c) forming a coupling with corresponding threads formed on the box portion and pin portion which satisfy the torque requirements as determined in the selecting and determining steps.

10. A sucker rod coupling including a threaded pin portion and correspondingly threaded box portion for use in connecting sucker rods having a rod outside diameter in a sucker rod string used to drive downhole pumping mechanisms to pump fluids from an underground formation to the surface comprising:

(a) a tapered, threaded pin formed on an end of a sucker rod to be coupled and a correspondingly tapered, threaded box for coupling with the pin which, when coupled have a coupling outside diameter no greater than about one and one half times the rod outside diameter of the sucker rod to be coupled and a coupling inside diameter no greater than the square root of the coupling outside diameter squared minus the square of the rod outside diameter, each of the box and pin having threads having a thread form that satisfies the torque requirements for the mated coupling wherein the torque requirements are defined by calculating the coupling yield interference using a thick wall pressure vessel as the mathematical model for the calculated interference to determine radial and tangential stresses for the coupling.

11. The connection of claim 1 wherein the box has a terminal end and wherein the connection includes an entrance defined by the terminal end of the box and further comprising a seal between the box and pin adjacent the entrance to prevent migration of production fluids into the mating threads.

12. The connection of claim 6 wherein the box has a terminal end and wherein the connection includes an entrance defined by the terminal end of the box and further comprising a seal between the box and pin adjacent the connection entrance to prevent migration of production fluids into the mating threads.

13. A threaded box and pin connection for connecting sucker rods having a rod outside diameter in a sucker rod string used to pump fluids from an underground formation to the surface comprising:

(a) a pin formed on an end of a sucker rod to be coupled and having external threads formed on a tapered core having a core diameter, wherein the core has a proximal end and a distal end and wherein a thread length and a taper are defined therebetween, wherein the ratio of the thread length to a largest diameter of the core is between .9 and 6.5 and wherein the taper, defined by the ratio of the core diameter at its proximal end minus the core diameter at its distal end to an axial length between the core diameters, is between .1 and .44; and

(b) a box having a correspondingly threaded and tapered counterbore for coupling with the pin, wherein an outside diameter of the mated coupling corresponds to an outside diameter of the box, which outside diameter of the box is no more than about one and one half times the core diameter at its proximal end, which corresponds to the rod outside diameter of the rod to be coupled.

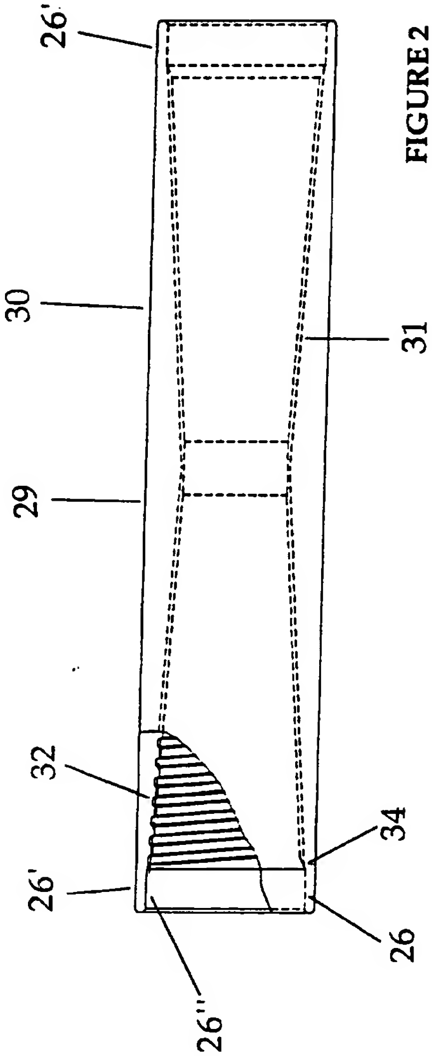

FIGURE 2

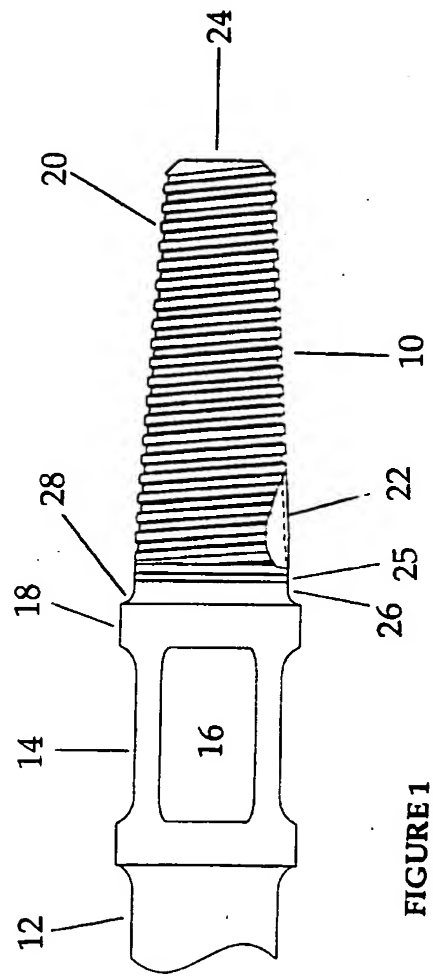

FIGURE 1

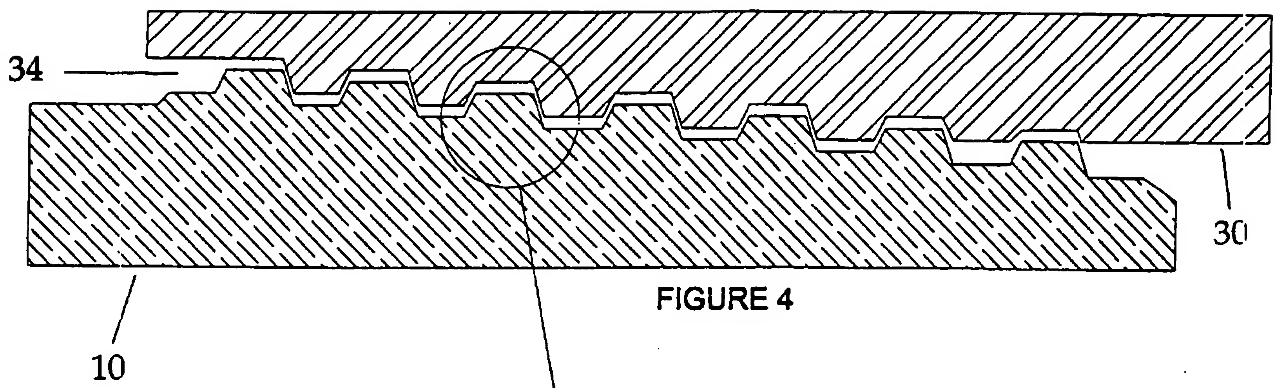

FIGURE 4

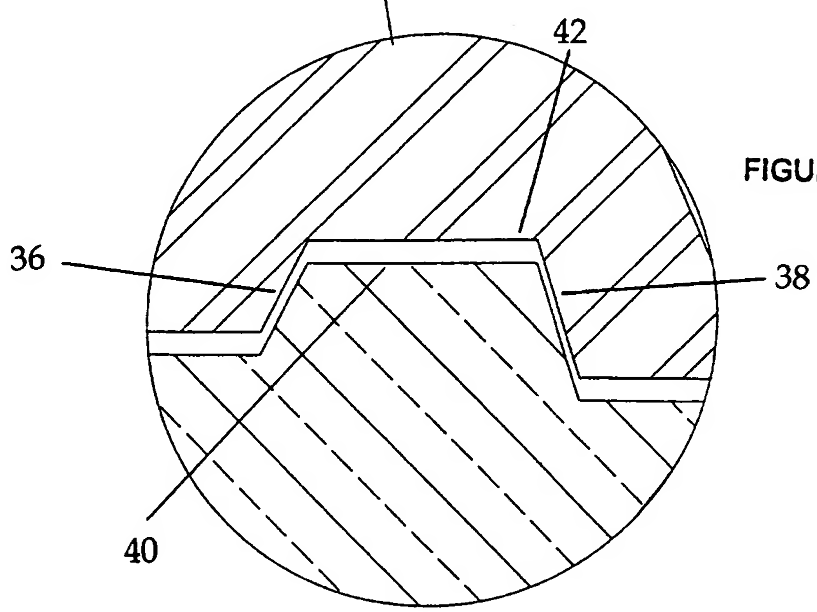

FIGURE 5

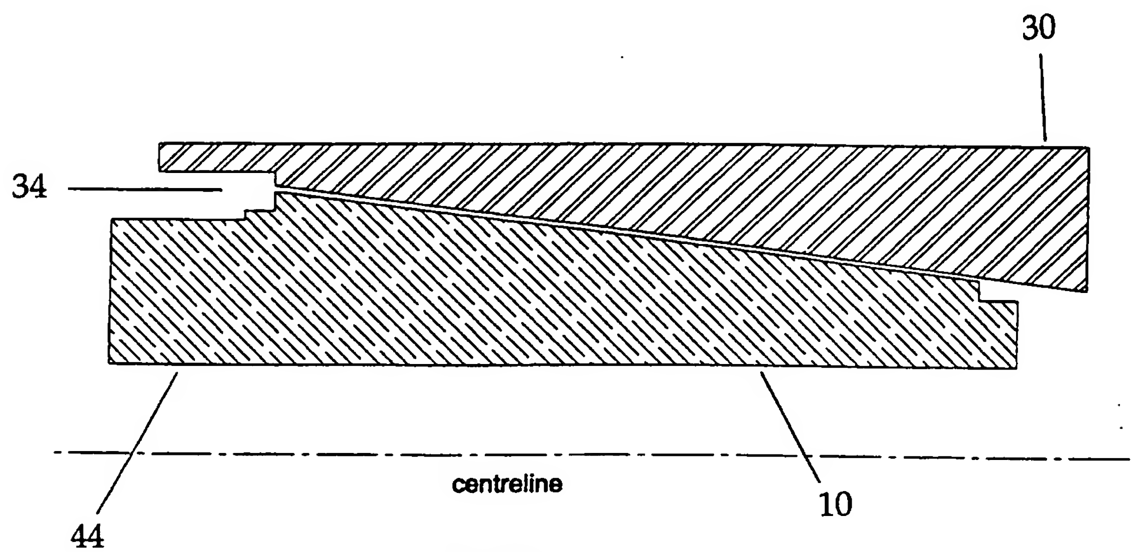

FIGURE 3

1.00x0.125

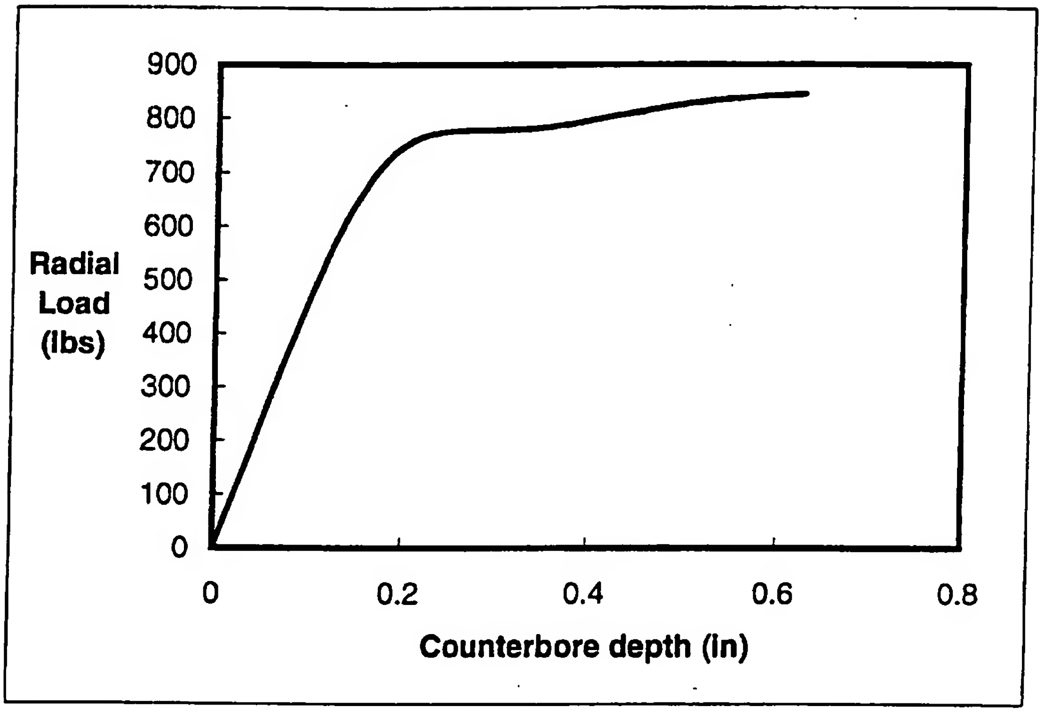

FIGURE 6

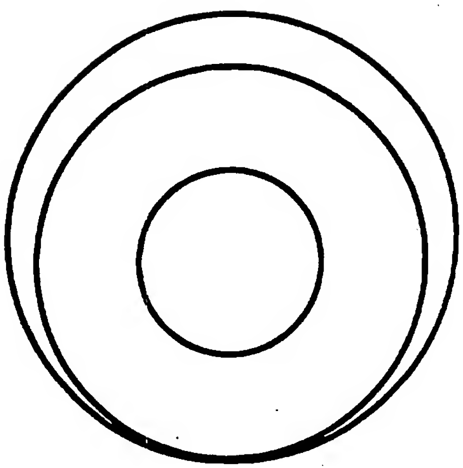

FIGURE 7A

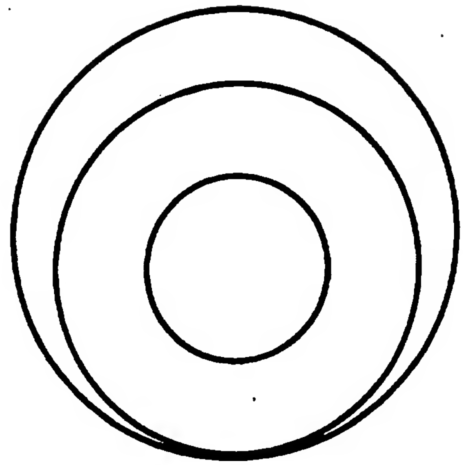

FIGURE 7B

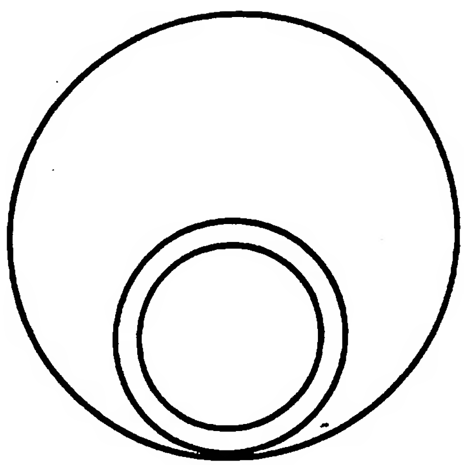

FIGURE 7C

FIGURE 7